

THESIS FOR THE DEGREE OF LICENTIATE OF ENGINEERING IN
MACHINE AND VEHICLE SYSTEMS

Dual mass flywheels in truck powertrains
Modelling, simulations and validation

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CHALMERS UNIVERSITY OF TECHNOLOGY

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Cover:

A picture and sketch of a dual mass flywheel, along with a heavy-duty truck powertrain.

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ABSTRACT

The heavy-duty vehicle manufacturers face large challenges when it comes to reducing CO₂ emissions from vehicles. The on-going development of more efficient combustion engines leads to an increase in torsional vibrations. In the future, the conventional flywheel and clutch will probably not be enough to protect the gearbox and rear driveline from engine induced vibrations. More advanced technology will be needed.

Dual mass flywheels (DMFs) have been used in smaller vehicles for many years and have shown to reduce the torsional vibrations transmitted to the gearbox. The use in heavy-duty commercial applications is to date very limited.

The work presented in this thesis focuses on DMFs for heavy-duty applications. It comprises modelling, measurements, correlation, development of numerical algorithms and complete powertrain simulations.

Two different DMF simulation models are used. The first one is a piecewise linear model, without the internal DMF parts explicitly modelled. It is used together with the harmonic balance method to evaluate resonances. The simulated results show that with piecewise linear DMF design, sub-harmonic resonance excitation can occur in operating speed range.

The second model includes the DMF internal parts. A simulation method where the dynamics problem is reformulated as a linear complementary problem (LCP) is proposed. The model is correlated based on test rig measurements on a DMF for heavy-duty applications. It is shown that the general DMF behaviour, as observed in the measurements, can be reproduced in the simulations for the speed and torque ranges studied.

The torsional vibrations in a heavy-duty truck powertrain with a single mass flywheel model and with the second DMF model are evaluated with simulations. The effects on resonance modes and frequencies when changing different powertrain parameters are presented. The simulations show that the vibration amplitudes are generally lower with a DMF. Resonance excitation can occur in operating speed range with the DMF and the DMF and clutch properties need to be adapted to the powertrain in order to obtain good vibration isolation in the complete operating speed and torque range.

Keywords: DMF, dual mass flywheel, torsional vibrations, heavy-duty truck, resonances, friction, simulations, powertrain, driveline

PREFACE

The work in this thesis has been carried out from May 2016 to November 2018 at the Department of Mechanics and Maritime Sciences at Chalmers University of Technology.

The research is part of the project "Reduced vibration transmissions - reduced energy consumption and environmental impacts together with an increased competitiveness". It is a cooperation project between AB Volvo, Scania CV AB, Chalmers University of Technology and KTH Royal Institute of Technology. The project is funded by the Swedish Energy Agency (project No. 42100-1), AB Volvo and Scania CV AB. The project includes four work packages with focus on clutches, dual mass flywheels, centrifugal pendulum absorbers and power split absorbers, respectively.

The thesis presents the work within work package 2, Dual Mass Flywheels. The theoretical work has been performed by me, in cooperation with my supervisors Professor Viktor Berbyuk, Associate Professor Håkan Johansson (both at Chalmers University of Technology) and Dr. Anders Hedman (AB Volvo). The experimental part has been performed as a master thesis work by Johan Karlsson under my supervision and with support from Johan Jonsson (AB Volvo), Anders Hedman and other personnel at AB Volvo.

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At last, I also want to acknowledge the contributions by Johan Karlsson and Johan Jonsson, regarding the experimental results. Their well performed work is greatly appreciated and valuable for the project.

Göteborg, October 2018

Lina Wramner

THESIS

This thesis consists of an extended summary and the following appended papers:

- Paper A** L.Wramner. Torsional vibrations in truck powertrains with dual mass flywheel having piecewise linear stiffness. *Proceedings of European Nonlinear Oscillation Conference, (ENOC 2017)*, Budapest, Hungary, (2017).
- Paper B** L. Wramner, V. Berbyuk, H. Johansson. Vibration dynamics in non-linear dual mass flywheels for heavy-duty trucks. *Proceedings of International Conference on Noise and Vibration Engineering, (ISMA 2018)*, Leuven, Belgium, (2018), 1863-1876.
- Paper C** L. Wramner. Numerical algorithms for simulation of one-dimensional mechanical systems with clearance-type non-linearities. *Submitted for international publication.*
- Paper D** L. Wramner. Torsional vibrations in heavy-duty truck powertrains with dual mass flywheels. *Submitted for international publication.*

The appended paper B was prepared in collaboration with the co-authors. The author of this thesis was responsible for the major progress of the work including taking part in the planning of the paper, developing the theories and the numerical implementation, performing the numerical simulations and writing the paper.

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Part I

Extended summary

1 Introduction

1.1 Background and motivation

Global warming is today considered by most scientists to be a serious threat to our environment. In order to prevent a significant global temperature rise, it is necessary to significantly reduce the emissions of greenhouse gases.

Many of the countries in the world have, with the Kyoto protocol and Paris agreements, committed to contribute to the reduction in the emissions. Within the European Union (EU) there is a target to cut greenhouse gas emissions by at least 40% below 1990 levels by 2030, [3]. Some individual countries have more ambitious targets.

Trucks, buses and coaches produce around a quarter of all CO₂ emissions from road transport in the EU and around 6% of the total CO₂ emissions in the EU, [4]. To significantly decrease the CO₂ emissions from the heavy-duty transport sector is a large challenge. Some reduction can be obtained by improved logistics, optimised vehicle use and when possible a shift to more ship and rail transportation. If not followed by an extensive reduction in the amount of transportation, these measures will most certainly not be enough to reach the targets.

Significant improvements in vehicle efficiency will also be needed. On May 2018, the European Commission presented a legislative proposal setting the first ever CO₂ emission standards for heavy-duty vehicles in the EU. The targets proposed for average CO₂ emissions from new trucks are 15% lower in 2025 than in 2019 and at least 30% (indicative target, subject to review in 2022) lower in 2030 than in 2019, [4].

For lighter vehicles in city traffic, electrification or hybridisation are promising measures for reaching the targets of reduced CO₂ emissions. For heavy-duty long-haul applications it is more complicated. The efficiency gain is not expected to be as high and the required size and weight of the batteries limit the space and weight available for cargo. Moreover, the capacity requirements on the batteries lead to very high battery costs. It is foreseen that combustion engines (with diesel or alternative fuels) will be needed for heavy-duty commercial applications in many years to come.

Reduced friction, improved aerodynamics and better engine and gearbox control can contribute to efficiency improvements, but it is also necessary to improve the efficiency of the engines. Some ways are to increase the cylinder pressure and to reduce the engine speeds, i. e. down-speeding the engines. Improved engine efficiency can also be obtained by optimising the engine for a smaller speed range. In that case, the gearbox ratio needs

to be adjusted to the vehicle speed more often in order to keep the engine speed within the desired range.

Many of the measures that can be taken to improve engine efficiency lead to increased torsional vibrations in the powertrain. This in turn has a negative impact on component life, comfort and emitted noise. Currently, there is also a higher focus on reducing the noise. In 2014, the EU adopted a new regulation with the purpose to reduce traffic noise by around 25%. The new regulation includes a new test method to better reflect current driving behaviour and a decrease of the limit values. For heavy-duty vehicles, the reduction will be 1 dB(A) until 2021 and 2 dB(A) until 2025, [5].

In the future, the vehicle manufacturers need to meet the increasing demand for higher engine efficiency while maintaining or reducing the levels of pollutant emissions and complying with the more strict noise regulations. This is not an easy task. Since engine improvements probably will not go without increased torsional vibrations, good isolation of the engine vibrations will be required. Today, a single mass flywheel (SMF) and clutch are used in most heavy-duty powertrains to protect the gearbox and rear driveline from the engine induced vibrations. In the future, with more efficient engines, it is foreseen that more advanced technology will be needed to limit the torsional vibrations transmitted to the driveline.

Dual mass flywheels have been used in passenger cars for decades and have shown to reduce the torsional vibrations. For heavy-duty applications, dual mass flywheels are not common and the potentials for such applications need to be better understood.

1.2 Aim of research

The aim of the research is to develop and validate simulation models for dual mass flywheels for heavy-duty applications and to use these models to simulate and evaluate the effect of introducing dual mass flywheels in heavy-duty powertrains. In particular, the following research questions will be in focus.

1. How can a DMF for heavy-duty applications be modelled?
2. How can torsional vibrations in powertrains with DMFs be simulated efficiently?
3. Which are the properties of the internal DMF friction and which are the proper friction model parameter values?
4. Which resonance phenomena can be expected if a DMF is used in heavy-duty commercial applications?
5. How are the torsional vibrations in heavy-duty truck powertrains affected when the conventional SMF is replaced by a DMF?
6. Can the use of a DMF enable engine down-speeding of heavy-duty commercial vehicles, without a significant increase in torsional vibrations?

2 Heavy-duty truck powertrains

A conventional heavy-duty truck powertrain is typically composed of a 4-stroke, 6-cylinder combustion engine, a single mass flywheel, a clutch, a gearbox, a propeller shaft, a rear axle and wheels as illustrated in figure 2.1.



Figure 2.1: Truck powertrain, source: AB Volvo (left) and schematic picture of powertrain (right)

Figure 2.2, shows typical maximum mean torque and power curves for a 13-liter engine. Combustion engines generate a highly non-uniform torque. In 4-stroke, 6-cylinder engines,

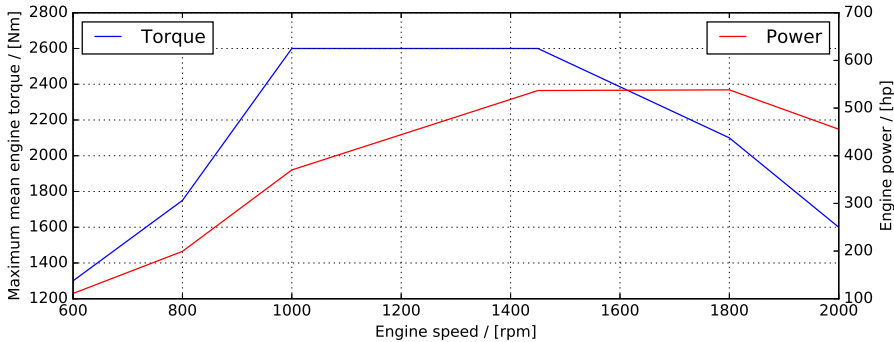


Figure 2.2: Typical heavy-duty engine torque and power (Volvo D13, 540 hp)

there are three cylinders firing each crankshaft revolution. This results in an oscillating torque from the engine, with three peaks each revolution and hence a frequency that depends on the engine speed. The frequency is referred to as the third order frequency. Figure 2.3 shows an example of how the torques at different powertrain positions vary during two crankshaft revolutions.

The SMF and clutch are designed to reduce the oscillating part of the torque transmitted from the engine to the rest of the powertrain. The SMF is basically a rigid wheel that provides isolation thanks to a large moment of inertia. The clutch also contributes

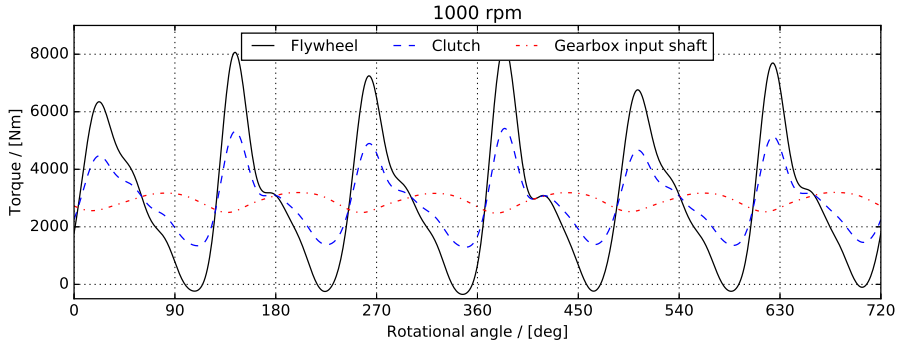


Figure 2.3: Torque at different powertrain positions

with moment of inertia. In addition, there is a damper in the clutch which provides a low stiffness torsional coupling between the clutch disc and the gearbox input shaft.

In typical heavy-duty truck powertrains, there is a resonance with frequency close to the third order frequency at low engine speeds. The resonance frequency depends on the gear selected and on the driveline configuration. Clutch damper design is very much focused on limiting the effect of this resonance. A decrease in clutch damper stiffness reduces the resonance frequency. Design limits on the clutch make it impossible in most cases to reduce the resonance frequency so much that the mode cannot be excited by the third order engine load in the operating speed range for all gears. Therefore the clutch damper also needs to provide significant friction damping in order to limit vibration amplitudes near resonance. The friction damping results in power losses and also leads to increased torsional vibrations at speeds far from resonance.

If the operating speed range of the engine is decreased, the resonance is expected to be more problematic.

3 Dual mass flywheels

Dual mass flywheels, as a means to reduce torsional vibrations in powertrains, started to emerge on the automotive market in the 1980s. Initially, DMFs were mainly used in top range passenger cars with powerful engines and high requirements on comfort. Today, almost half of all vehicles produced with manual gearshift transmissions are equipped with a DMF [10]. In commercial heavy-duty vehicles, conventional single mass flywheels (SMFs) are still standard and the use of DMFs is mainly restricted to buses and coaches.

A DMF is composed of two flywheels, torsionally connected by springs. By exchanging an SMF for a DMF, the powertrain resonance frequencies are shifted. It is generally possible to design the DMF so that the DMF resonance speed is well below the idle speed of the engine [8]. This leads to very good vibration isolation at low engine speeds.

In figure 3.1, the two most common types of DMF design are illustrated schematically. With the arc spring design in 3.1a, bent springs are located in a toroid-shaped channel in the primary flywheel. When there is relative angular motion between the two flywheels, the springs are compressed and slide against a guide in the channel. With the straight spring design in 3.1b, the two flywheels are instead connected by several straight springs in series, with sliding shoes in between.

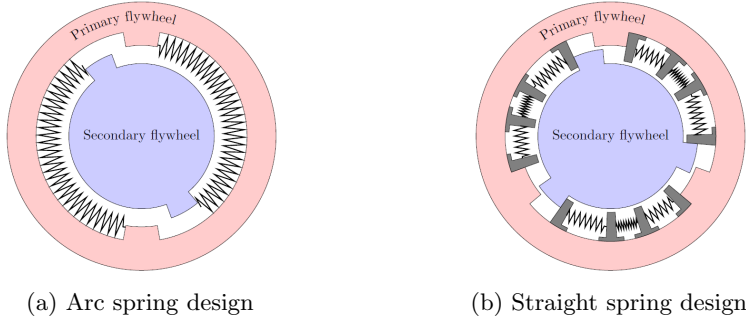


Figure 3.1: DMF types

There are many variants of the above two DMF types. The number of springs and stiffness of the springs vary, and the springs generally need to be optimised for different applications. Usually there are many springs with different stiffness and lengths inside each other. Low stiffness is generally good from a vibrational perspective, but a lower stiffness also reduces the maximum torque that can be transmitted. The DMF is often designed to have a piecewise linear stiffness. At low mean torques, the stiffness is low, leading to low vibration amplitudes. At high mean torques, the stiffness is high, which enables the DMF to transfer high torques without bottoming.

In DMFs with the arc spring design, relative motion between the two flywheels results in friction forces between the guide in the primary flywheel and the springs. In DMFs with straight spring design, there is friction between the sliding shoes and the primary flywheel. The normal force at the friction surfaces depends on the compression of the springs and on the centrifugal action. With high vibration angles and high mean torques, the friction is higher, [8, 11, 10]. Moreover, at higher speeds, the springs are compressed inhomogeneously, leading to a shorter effective length of the springs and hence a higher effective stiffness [11].

These characteristics of the DMF result in high damping and low stiffness at start-up when the resonance is passed. At normal driving, with low relative vibration angles, there are generally low frictional losses, [1, 8].

4 Models and methods

Simulations have been the core of this work. Today, simulations are used extensively within the industry, in all phases of product development. Reliable simulation models and methods are therefore very important.

The engine operates in a wide torque and speed range and what is a good powertrain design with respect to one load scenario, might be a bad design with respect to some other load scenario (see **Paper D**). It is therefore often necessary to simulate a wide range of different load cases. If parameter studies or optimisations are to be performed, even more simulations are needed. Therefore, it is not enough with accurate and reliable simulations. The simulation times must also be kept low.

The work presented in this thesis comprises modelling, simulations and validations.

The purpose with the simulations has been to evaluate engine excited torsional vibrations at high load in heavy-duty truck powertrains with an SMF and with a DMF. Focus has been on the vibrations transmitted from the engine to the gearbox. In particular, frequencies in the range of 30 Hz - 100 Hz are studied. These frequencies correspond to the third engine order frequencies from idle speed to maximum engine speed. It is usually for frequencies in this range that the torque amplitudes into the gearbox reach the highest values. The simulated results are presented in **Paper A** and **Paper D**.

The models used in the simulations are developed with the purpose to be computationally efficient while still providing good accuracy for the analyses presented.

It was decided to only simulate torsion, since the coupling between the torsional dynamics and the translational dynamics in truck powertrains in the load and frequency range studied usually is relatively small. This simplification is also common in literature for analyses of powertrain torsional dynamics in smaller vehicles and has shown to give good correlation with measurements, [7, 2, 15, 12, 9].

Torsional lumped mass models with up to 27 degrees of freedom representing the complete powertrain from engine to wheels have been used in the simulations. Details about how the modelling has been done can be found in **Paper A** and **Paper D**. The models are selected to be representative for a typical heavy-duty truck. Shapes of the lowest resonance modes, with frequencies in the range in focus, are for most heavy-duty truck powertrains very similar. The frequencies of the modes can vary, and as a result we can see resonance excitation occurring in different trucks at somewhat different engine speeds.

In literature, it is common to model the rotational parts of the engine as one lumped mass, ([7, 2, 14, 13, 6]). For standard flywheels with large moment of inertia, this is in most cases a good approximation that simplifies implementation and reduces simulation time, without a significant impact on the results. Early in this project, while evaluating different modelling approaches, it was observed that near resonances, the engine model can significantly influence the results rear of the flywheel. Since resonances are studied both in **Paper A** and **Paper D**, it was decided not to model the engine as one lumped

mass. Instead an engine model which includes the flexibility of the crankshaft has been used. Each crank pin is represented by a lumped mass at which the loads from the cylinders are applied.

In the full powertrain simulations, presented in **Paper D**, non-linear models are used for the DMF and clutch, while the rest of the powertrain is treated as linear. For the resonance modes involved in the engine excited vibrations studied in this work, clutch and dual mass flywheels usually play a big role. The damping of these components therefore often has a large impact on the vibration amplitudes near resonances. A very accurate modelling of the friction is hence necessary to well predict the vibrations. Moreover, due to many internal springs with clearances and friction, the effective stiffness of the DMF can vary significantly with engine speed and load, [11]. Much work has therefore been put into the modelling of the DMF. The non-linear DMF model used is described in **Paper B** and **Paper D**. Test rig measurements have also been performed, in order to correlate the model and the friction parameters as described in **Paper B**. The damping in the rest of the powertrain is usually low or does not have a large impact on the steady-state engine excited torsional vibrations into the gearbox, studied in this work. Therefore, a linear modelling has been considered appropriate for the remaining components.

A low model complexity is one key to efficient simulations, but the selected computational method is also important. Several different simulation methods have been explored in this project. In **Paper A** and **Paper C**, some different computational algorithms are proposed. **Paper C** also has thorough comparisons and analyses of different computational methods and algorithm parameter settings. The effects of numerical damping and simulation time step on computational accuracy and convergence are also evaluated.

5 Summary of appended papers

Paper A: Torsional vibrations in truck powertrains with dual mass flywheel having piecewise linear stiffness. A complete powertrain model has been used in order to evaluate how a piecewise linear DMF design affects the vibrations in the powertrain. Simulations have been performed with the harmonic balance method. Evaluation is done both with respect to mode shapes and frequencies and computed steady-state vibration amplitudes. In the linear region, there is a frequency shift for a problematic resonance mode that leads to a significant decrease in vibration amplitudes at low engine speeds. In non-linear regions, a resonance mode with frequency corresponding to half the main exciting frequency from the engine can be excited, leading to high vibration amplitudes. The frequency of this mode and the extent to which it is excited depend on the engine torque, and the highest amplitudes are not always obtained at the highest load.

Paper B: Vibration dynamics in non-linear dual mass flywheels for heavy-duty trucks. In the paper, a non-linear model for simulation of a dual mass flywheel for heavy-duty applications is proposed. The model includes internal clearances and friction. The LuGre friction model is used, which depends on normal force, relative velocity between the two surfaces and an internal deflection variable. Measurements on the DMF are performed in a test rig and the test rig properties are analysed. The correlation shows that the general behaviour of the DMF is reproduced by the simulation model proposed. The viscous part of the friction is dominant for the cases analysed, with zero mean torque, and a Coulomb friction model would not suffice for this application.

Paper C: Numerical algorithms for simulation of one-dimensional mechanical systems with clearance-type non-linearities. Generalisations of an efficient algorithm for simulating systems with gap activated springs connecting bodies are proposed in this paper. The simulation problem is reformulated as a Linear Complementary Problem (LCP) for which there are very efficient solutions. The generalisations enable the LCP approach to be used for an arbitrary number of gap activated springs connecting either different bodies or connecting bodies to ground. The springs can be activated in either compression or expansion or both and a gear ratio can be included between the bodies. The efficiency of the algorithm is verified with an application example representing a dual mass flywheel. It is also concluded that if numerical damping is used, the time step can be reduced significantly when simulating a DMF with many clearances, without large penalties on accuracy.

Paper D: Torsional vibrations in heavy-duty truck powertrains with dual mass flywheels. In this paper, complete models of heavy-duty truck powertrains with SMFs and DMFs are used to evaluate the torsional vibrations. The focus is engine excited vibrations at high load within operating speed range. The torsional resonance frequencies and modes are presented and it is described how different powertrain parameters affect the resonances. Results from simulations of steady-state torsional vibrations in heavy-duty truck powertrains with SMFs and with DMFs are shown and evaluated. It is demonstrated how a proper choice of DMF and clutch properties can enable down-speeding and increased engine torque without severe impact on the torsional vibrations into the gearbox.

6 Conclusions and outlook

6.1 Conclusions

This work has been focused on modelling, simulations and validation of DMFs in heavy-duty truck powertrains. To conclude, the research questions in section 1.2 are recaptured and answered briefly.

1. How can a DMF for heavy-duty applications be modelled?

Two different levels of DMF modelling are used in this work. In **Paper A**, a piecewise linear model, without the internal parts of the DMF explicitly included, is described and used in the simulations. In **Paper B**, a detailed model of a straight spring heavy-duty truck DMF is proposed. The model includes the internal springs and sliding shoes and considers internal friction and clearances. The necessary modelling complexity depends on the purpose with the simulation and on the accuracy required. Both models capture powertrain resonances in a realistic way, including the sub-harmonic resonance excitation from the non-linear DMF stiffness. In order to have realistic damping levels in a larger speed and torque range, the more detailed model is necessary, as can be seen in **Paper B**.

2. How can torsional vibrations in powertrains with DMFs be simulated efficiently?

The proper choice of simulation method depends on the simulation model and goal with the simulation. In **Paper A** it is shown how the harmonic balance method can be used with the piecewise linear DMF model to efficiently simulate steady-state torsional vibrations. The harmonic balance method solves in frequency domain and is also suitable for evaluating non-linear normal modes which can be a help in understanding resonance behaviour. In **Paper C** it is proposed how systems with many gap-activated springs can be modelled as a linear complementary system which can be efficiently solved. The effect of introducing numerical damping in the algorithm is also evaluated. It is concluded that if numerical damping is used, the time step can be reduced significantly when simulating a DMF with many clearances, without large penalties on accuracy.

3. Which are the properties of the internal DMF friction and which are the proper friction model parameter values?

The measurements described in **Paper B** show that there is significant viscous damping in the DMF and that the viscous component must be included in the friction model. By including significant viscous damping between the sliding shoes and the primary flywheel, the general DMF behaviour, as observed in the measurements, could be reproduced in simulations. Realistic friction parameters for the DMF and the operating range studied with zero mean torque are presented in **Paper B**. Future work is needed to verify the model also for high mean torques.

4. Which resonance phenomena can be expected if a DMF is used in heavy-duty commercial applications?

The use of a DMF affects the resonance modes and frequencies in the powertrain. In conventional heavy-duty truck powertrains there is a resonance, with large motion in the clutch damper, that can be excited by third engine order at low engine speeds in the operating speed range, as discussed in **Paper A** and **Paper D**. When the SMF is replaced by a DMF, a new mode, referred to as mode B in **Paper D**, is introduced with large motion in the DMF springs. The DMF can be designed so that the frequency of this mode coincides with third order frequency well below idle speed. If the DMF has piecewise linear characteristics, the mode can be excited when the third order frequency is twice the resonance frequency. It is only excited for some mean torques and the highest excitation is not necessarily at the highest torque, as concluded in **Paper A** and shown in **Paper D**. There is also a mode with large motion in the clutch damper in the DMF powertrains, referred to as mode C. The frequency of the mode depends on the stiffness of the clutch damper. With the typical heavy-duty truck powertrains studied in **Paper A** and **Paper D**, the frequency of this mode corresponds to third order frequency in operating speed range.

5. How are the torsional vibrations in heavy-duty truck powertrains affected when the conventional SMF is replaced by a DMF?

In general, the introduction of a DMF provides good vibration isolation at low engine speeds, thanks to the low resonance frequency of mode B, as shown in **Paper D**. With the DMF powertrain studied in **Paper D**, excitation of mode C can occur for high mean torques and low gears in the operating speed range, leading to high vibration amplitudes. Excitation depends very much on the friction in the clutch, and by adjusting the clutch friction the high vibration amplitudes near resonance speed can be avoided.

6. Can the use of a DMF enable engine down-speeding of heavy-duty commercial vehicles, without a significant increase in torsional vibrations?

As discussed in **Paper D**, a well matched combination of DMF and clutch can provide very good vibration isolation at low engine speeds and thereby enable down-speeding without significant increase in torsional vibrations. The properties of the DMF and clutch must be adapted to the powertrain in order to avoid large resonance excitation in operating speed range, and all gears need to be considered. A compromise between good properties at low and high mean torques is generally needed.

6.2 Recommendations for future work

In the experimental parts of this work, the vibrations in a heavy-duty truck DMF with various torque amplitudes and frequencies at a zero mean torque were analysed. The mean torque affects the normal force at the friction contact surfaces. Therefore additional measurements are required to validate that the friction modelling is appropriate for the complete DMF operating range and that friction model parameters are realistic.

The powertrain analyses presented in this thesis are focused on steady-state vibrations and powertrain resonances at high engine loads and with the clutch engaged. Transient vibrations occurring at start-up, gear shifting and stalling can reach very high amplitudes. These type of vibrations should be subject to future work.

In the future, new engine technologies will most certainly be introduced that will affect the vibratory behaviour of the engine. One example is cylinder deactivation. By deactivating cylinders, the efficiency of the engine at partial load levels can be increased. As a consequence, the frequencies of engine load oscillations are reduced which can lead to resonances being excited at higher engine speeds. How this will affect the vibration amplitudes in heavy-duty drivelines with and without a dual mass flywheel needs to be investigated further.

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